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Experimental Determinations of the Flow Characteristics in the Volutes of Centrifugal Pumps

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This paper is a discussion of an exhaustive study which was made to determine internal-flow characteristics of two high-efficiency, high-head centrifugal pumps of commercial design. Special equipment was constructed to measure instantaneous values of pressures and velocities at various stations in the volutes. The authors discuss previous work of a nature similar to their study. They describe briefly the apparatus used in their tests, the arrangement of the test equipment, and the accuracy of the equipment used to measure the instantaneous values of pressures and velocities in the volutes. The results of the tests are presented in graphical form.

Background and Objectives

THE DESIRE for a knowledge of actual flow conditions in the impellers and volutes of centrifugal pumps has long been present in the minds of those interested in the construction or operation of hydraulic machinery. This desire has grown more intense as efficiencies have been forced higher and higher, since each increase in efficiency has been more difficult to attain, and has demanded more precise information about the hydraulic behavior within the casing of the machine. Therefore, it is not surprising that many attacks have been made on this problem, both from the theoretical and the experimental points of view. However, as yet neither method of approach has yielded entirely satisfactory results, so that additional attempts to supply this knowledge are still definitely needed.

Theoretical Treatments. Many of the theoretical studies have started with the assumptions of perfect fluids and potential flow. The works of Kucharski, (1)[3] von Busemann, (2) Schultz, (3) Sörenson (4), and Uchimaru and Kito (5) are in this class. Kucharski treated mathematically the problem of an impeller with straight radial vanes. Spannhake (6) pointed out that fluid passages, formed by curved vanes of finite length and cut off by entrance and exit circles, as found in actual practice, present many difficulties to theoretical investigators. It is well known that the actual values of both the magnitude and direction of the absolute exit velocity do not agree with those calculated on the basis of potential flow. Pfleiderer (7) therefore calculated the theoretical head developed by a pump on the condition that the relative exit angle was less than the vane angle. Fischer and Thoma (8) concluded that: "Practically all flow conditions for an actual fluid are fundamentally different from those theoretically derived for an ideal frictionless fluid."

It should be noted that the treatments just mentioned refer almost entirely to the flow within the impeller and give few or no data on the action within the volute. This is decidedly a shortcoming, since a large part of the energy delivered to the fluid by the impeller is discharged from it in the form of kinetic energy, and must be transferred to pressure energy in the volute. Therefore, a knowledge of the flow conditions in the volute is very desirable.

Daugherty (9) has combined a theoretical analysis with a study of the actual performance characteristics of certain pumps. One result of particular interest is his conclusion that the vane angle and the actual relative exit angle may differ by as much as five to ten degrees.

Experimental Investigations. Probably one of the first experimenters to study the flow conditions inside an actual rotating hydraulic machine was Francis (10) in 1851. In his Tremont turbine test he inserted a vane in the runner discharge, which gave the direction of the water leaving the wheel.

Photographic studies have been made by Fischer and Thoma, (8, 11) Oertli, (12) Stiess, (13) Closterhalfen, (14) and others. Fischer and Thoma worked with a pump having an open impeller and a glass side. The flow was made visible by dye injections at various points in the impeller passages. This was studied by use of a rotating prism which made the impeller appear to stand still, and was photographed by a rotating camera. Closterhalfen also used a pump with a transparent case, and in addition measured the pressures at some points along the vanes. Oertli showed that the flow in the impeller was not exactly two-dimensional.

All of these studies were carried out on pumps especially constructed for the purpose, with the design modified to permit of radial plane windows and other necessary modifications. The heads, capacities, efficiencies, and Reynolds' numbers were all low. The two latter indicated that there is good reason to expect a difference between characteristics of flow found in these pumps

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

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and those existing in modern large, high-efficiency machines. Nevertheless, these studies have been very valuable in pointing out the discrepancies between the present theoretical treatments and the actual flow conditions, and also in showing the way to carry the work further.

It should be mentioned that Yendo (15) used pressure-measuring holes in the guide vanes of a turbine pump to obtain the slip coefficient. However this method does not give a measurement of the magnitude of the impeller exit velocity.

The first study of internal-flow characteristics to be undertaken in the hydraulic laboratories of the California Institute of Technology was initiated in the fall of 1931 by the present authors. The development of both instruments and technique had reached such a state by the spring of 1933 that a master's thesis was presented by Binder (16) on an investigation of the flow characteristics in the volute of one of the laboratory pumps.

In 1936, Kasai (17) reported his studies of 1933 and 1934. His instruments and method of attack followed closely those outlined in Binder's thesis (16). Although Kasai also used a specially constructed pump which had a vortex chamber between the impeller and the volute, it was of a reasonable capacity and had a good efficiency. Therefore, it is felt that his work represented a definite advance over the investigations previously discussed.

Expansion of Facilities. In the fall of 1933, the design and construction of a new hydraulic-machinery laboratory was started under a cooperative agreement between the Metropolitan Water District of Southern California and the California Institute of Technology. A description of this laboratory and its equipment has been given by Knapp (22). The program of investigation of this laboratory offered exceptional incentives to continue the work already started on the internal-flow conditions. The laboratory equipment available was adapted very well to such a study, as it offered means for exact control of all test conditions, and instruments for making precision measurements of the pump performance. A group of pumps from different manufacturers were available upon which this investigation could be carried out. They had all been selected carefully to represent the best practice in efficiency and general performance. They were of sufficient size (7 and 8 in. discharge) so that results obtained from them could be considered typical for high-head high-capacity units. In addition to having such satisfactory facilities available, it was felt that the severe conditions under which the Aqueduct pumps would operate necessitated a thorough knowledge of the internal-flow characteristics in order to insure maximum efficiency and trouble-free operation. Therefore, it was decided to proceed with the investigation which is the main subject of this article.

Objectives. The chief experimental objectives of this study have been to obtain a complete analysis of both instantaneous and average values of pressures and velocities in the volutes of the pumps investigated.

It was felt that a knowledge of the instantaneous velocities close to the impeller discharge, together with measurements of their variation with phase, i.e. with the relative position of the impeller passage, would prove very valuable in analyzing the flow in the impeller itself, since from such measurements the velocity distribution at the discharge end of the impeller passages can be calculated.

The knowledge of the average values of velocity gave promise of being useful both in aiding to understand the flow in the volute itself, and in ascertaining the changes in flow conditions in a given impeller passage during each revolution as it discharged into successively different parts of the volute.

The pressure distribution, taken together with the velocity distribution, not only should help to explain the flow characteristics, but also should furnish a basis for calculating the radial forces acting on the impeller.

With this brief discussion of the objectives of the investigation as a background, a description of the methods and instruments used will be presented before the experimental results and the conclusions are offered for consideration.

Technique of Measurement

A general discussion of the experimental methods employed to measure velocity vectors will be given first, followed by a short discussion of each major instrument. No attempt will be made in the present paper to give all of the details of the technique used, but it is hoped that a more complete description will be presented in a subsequent article.

Fig. 1 is a diagram of the apparatus used. Briefly stated, the method employed was to insert a special direction-finding pitot tube across the volute. A sampling slide valve was inserted in each of the two connections from the pitot tube to the special differential gage. These slide valves opened for a short interval

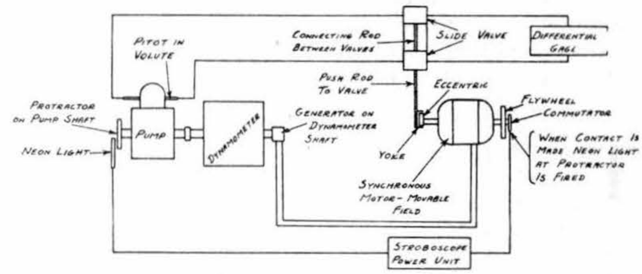


Fig. 1 Arrangement of Test Apparatus

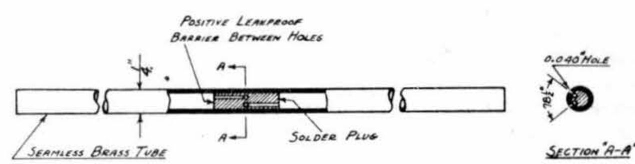


Fig. 2 Special Pitot Tube Used in the Tests
(Note: Holes drilled accurately in the same plane, normal to axis of tube.)

of time each revolution of the pump, which resulted in a series of pressure impulses to the gage. Means were provided for shifting the phase between the pump shaft and the valve opening.

A stroboscope indicated the position of the opening. Thus, the velocity could be measured as any particular point of the impeller passed the pitot tube.

The Pitot-Tube Measurement. Fig. 2 shows the pitot tube used, which is sometimes called a "direction-finding" pitot by wind-tunnel experimenters. Fig. 3 shows the method of insertion in the different pump volutes.

In any flow measurement it is not difficult to determine the true total head (velocity plus static), for the total head is obtained by placing an opening normal to the stream. However, for an accurate determination of velocity, it is also necessary to have a precise measurement of the static head, and this is much more difficult to secure. One of the main features of this pitot tube is that it gives an accurate measurement of static head in turbulent flow.

Considering the pressure distribution around a small cylinder across a stream, it is known that there is a critical angle with the direction of flow at which the velocity pressure has no effect. This means that, having an opening at the critical angle with the flow direction, the pressure transmitted to a gage will be truly static and unaffected by any influence of velocity.

Dryden (18) and Fechheimer (19) were the early contributors toward the development of this type of pitot tube for air measurements. Fechheimer found the critical angle to be $39^1/_4$ deg. The authors have checked this critical angle and the construction of $^1/_4$-in. and $^3/_{16}$-in. diameter pitot tubes by observing the position of the hole in a stream of known direction where the static pressure was known. This check gave an angle of $39^1/_4$ deg for the velocity range met in these pumps tests. It is interesting to note that these later measurements were made in water, but that the Reynolds number was substantially the same as that used by Fechheimer.

Referring to Fig. 2, the small pressure openings were possible because of the use of a special differential gage to be described later. In using the tube in the pump volute it was necessary to "balance" the tube. Each static hole was connected to one side of the differential gage. The pitot tube in the unknown stream was rotated about its own axis until the pressures at each hole were the same, in other words, the differential pressure was zero. At this position velocity pressure had no effect on either hole, and

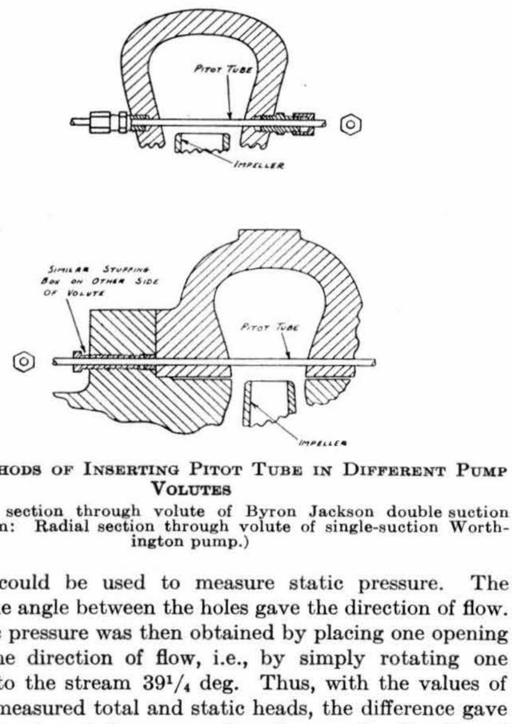


FIG. 3 METHODS OF INSERTING PITOT TUBE IN DIFFERENT PUMP VOLUTES

(Top: Radial section through volute of Byron Jackson double suction pump. Bottom: Radial section through volute of single-suction Worthington pump.)

either hole could be used to measure static pressure. The bisector of the angle between the holes gave the direction of flow. The dynamic pressure was then obtained by placing one opening normal to the direction of flow, i.e., by simply rotating one hole back into the stream $39^1/_4$ deg. Thus, with the values of the directly measured total and static heads, the difference gave the velocity head, and the measured angle gave the direction of the velocity vector.

An interesting and delicate technique was finally developed for constructing the pitot tube. After carefully tinning and then cleaning the inside of the brass tube, the holes were accurately drilled in a special jig. A piece of clean polished piano wire was inserted through one hole from the inside and extended out one end, while another piece of piano wire was arranged likewise for the other hole. A small metal plug was placed about $^1/_2$ in. below the plane of the holes and small pieces of solder filled in; heating carefully in an electric heater (for close temperature control) and in a reducing atmosphere (to prevent foreign matter from interfering with the perfect barrier) the solder plug, or barrier, was formed between the holes. Pulling out the piano wire as the solder solidified left the desired passages. The tube was then tested for the barrier, and checked for angle and velocity accuracy.

Special apparatus was devised for checking these pitot tubes. Using a free jet, many tests were made on the magnitude of velocity measurement, and very close agreements obtained for the range of high velocities met in pump traverses. Using this special apparatus with a closed jet, a wall correction was found which was applied to pump traverses. This wall-correction curve is shown in Fig. 4.

Special Differential Gage. An ordinary mercury or water U-tube manometer would be out of the question on these fluctuating pressure measurements. To obtain a reading in a system using an ordinary U tube, an appreciable flow is required through the pitot pressure openings and the connecting leads; hence, a

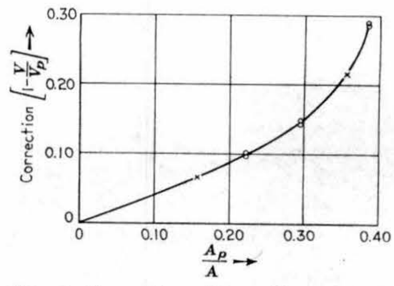


FIG. 4 PITOT-TUBE WALL CORRECTION

(A_p = projected area of the pitot tube, A = area of pipe without pitot tube, V_p = mean velocity indicated by pitot tube traverse, and V = velocity = quantity per unit time divided by A.)

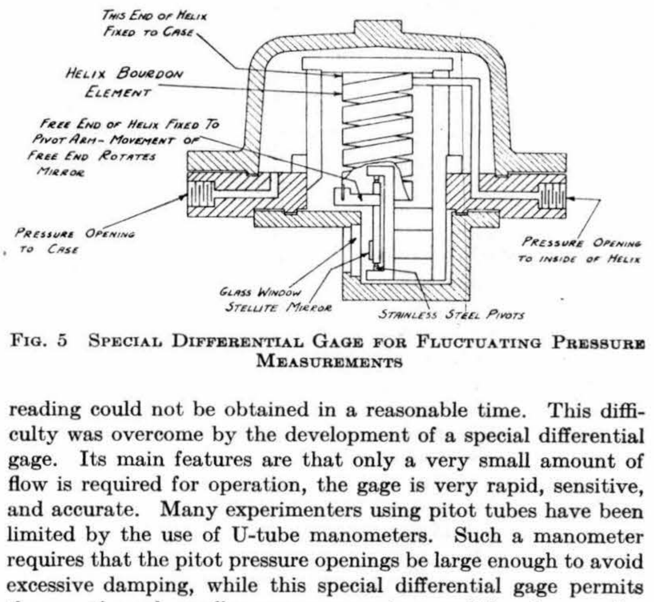


FIG. 5 SPECIAL DIFFERENTIAL GAGE FOR FLUCTUATING PRESSURE MEASUREMENTS

reading could not be obtained in a reasonable time. This difficulty was overcome by the development of a special differential gage. Its main features are that only a very small amount of flow is required for operation, the gage is very rapid, sensitive, and accurate. Many experimenters using pitot tubes have been limited by the use of U-tube manometers. Such a manometer requires that the pitot pressure openings be large enough to avoid excessive damping, while this special differential gage permits the use of much smaller pressure openings, and therefore smaller tube diameters.

Fig. 5 shows the internal construction of the differential gage. Since the helix element is the same as is used on pressure-recording instruments, this differential gage can be adapted to any desired accuracy and range of pressure by a suitable selection of helix. One end of the helix element is fixed, while the other end is free to move. The free end is so connected as to cause a rotation of the stellite mirror when the free end moves. Water pressure is applied to both the inside and the outside of the Bourdon element, the whole mechanism being in water in a closed case. Thus, when the differential pressure changes, the free end of the Bourdon element rotates the mirror. The mirror arrangement magnifies this movement with the aid of an optical system. A light source sends a beam of light through the glass window to strike the mirror. The reflected ray is focused on a graduated scale. The complete gage setup is shown in Fig. 6.

The gage was calibrated with a deadweight gage tester, and gave a straight-line calibration curve. Repeated tests over long periods of time have shown that this gage holds its calibration precisely. Tests have shown that the scale deflection depends solely on the differential pressure and is independent of the absolute pressure.

One interesting feature of this gage is that it requires no time to give a pressure reading. In this gage there is no appreciable flow of water, it is practically a constant-volume system.

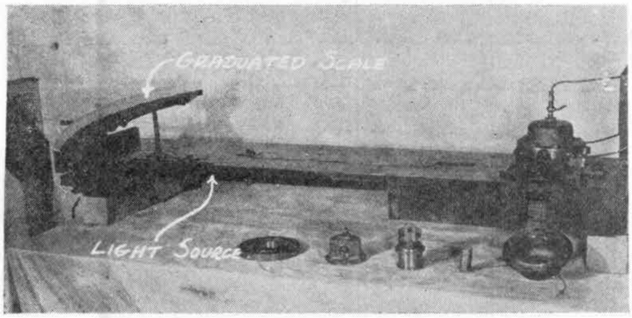


FIG. 6 SETUP OF THE DIFFERENTIAL GAGE

Sampling Valve and Phase Shifter. With the pitot tube and the special differential gage, measurements of average velocity can be made in the pump volute. This in itself is an improved technique, but further developments were made. Because of the extremely minute flow required to operate the differential gage it was possible to use a slide valve to sample the pressure transmitted from the pitot as any particular point on the impeller passed the pitot tube. The following description will show how this was accomplished.

Referring to Fig. 1, the generator on the dynamometer shaft drove the synchronous motor at one-half pump speed. An eccentric on the motor shaft worked in a yoke to impart simple harmonic motion to the push rod driving the two valves. Each valve opened twice (back and forth) for every revolution of the synchronous motor, which meant one valve opening per revolution of the pump.

When the valves opened, the commutator contact would fire the neon light at the protractor on the pump shaft (one firing per one revolution of pump). The bolts between the field and the end bells of the valve motor were removed, and means provided for rotating the field of the motor. Thus there was a positive mechanical-electrical connection between the pump shaft and the slide valve, and by simply rotating the field of the motor it was possible to change the phase relation between the pump shaft and the time of opening of the valve. It was possible to change the phase by 360 deg, while the stroboscope always gave a precise indication of the position of opening of the valve.

Fig. 7 shows the construction of one of the valves. E. R. Lockhart (20) helped work out the details of the sampling valves. The valves are duplicates, and accurately positioned to open at exactly the same time.

Use was made of the fact that the eccentric and yoke imparted simple harmonic motion to the valve, with the result that at the middle of the valve travel the velocity is a maximum while the acceleration is zero. The valve therefore was set to open at the middle of the travel. With a slot thickness of 0.005 in., the time of opening corresponds to an angular rotation of 5 deg of impeller. Tests have shown that the motion of the valve at any speed has no effect on the pressure transmitted through the valve. This is probably due to the fact that the valve opens at the point of zero acceleration. Whether or not this is the complete explanation, it is an experimental fact that the slide valve has no effect on the pressure transmitted.

One very interesting check was made. With the apparatus installed on a pump, in place of the pitot tube an oscillating pressure of known frequency was applied to the slide valve. The pump was run at various speeds, each different from the known frequency of the applied pressure. For each case the number of "beats" per minute, as shown by the differential gage, corresponded exactly to the difference between the pump speed and the cycles per minute of the applied pressure.

Tests have shown that each slide valve when closed does not leak. A typical installation is shown in Fig. 8.

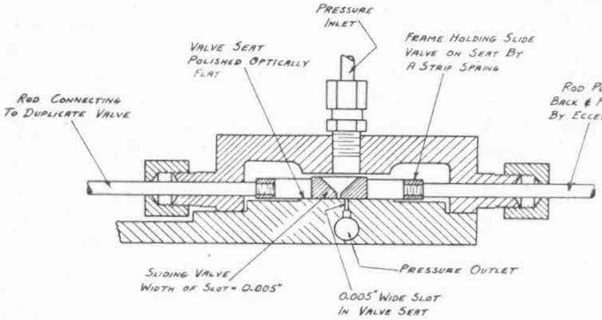


FIG. 7 CONSTRUCTION OF SLIDE VALVE

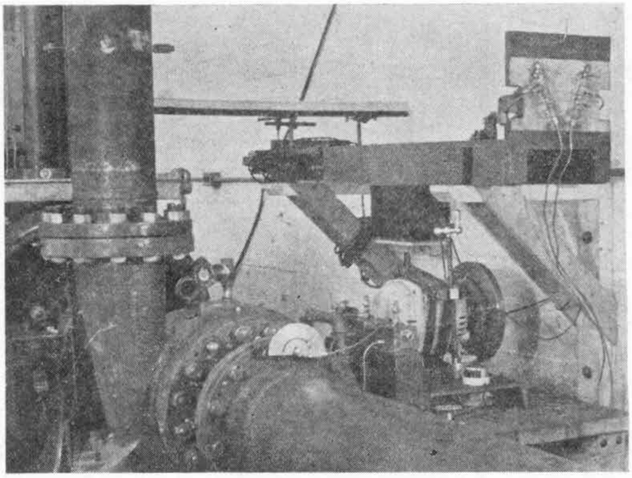


FIG. 8 A TEST INSTALLATION

General Remarks on Technique of Measurement. This technique of instantaneous velocity measurement is possible because the system from pitot tube to gage has practically no volume changes; the pressure is transmitted by the water without any appreciable flow. Pressure waves in the connecting leads might cause trouble, but the length of the leads was reduced to a minimum by placing the slide valve as close as possible to the pump.

PRESENTATION OF TEST RESULTS

Notation Used in Measurements. For designating valve opening, one vane-tip edge was chosen as a zero reference. If the valve opened as the zero reference mark passed the pitot tube the "phase angle was 0 deg." If the valve opened as some other point on the impeller passed the pitot tube, this point was referred to the zero mark as so many "degrees phase angle," where this angle is measured in the opposite sense to that of the pump rotation, i.e., the point lags the zero reference.

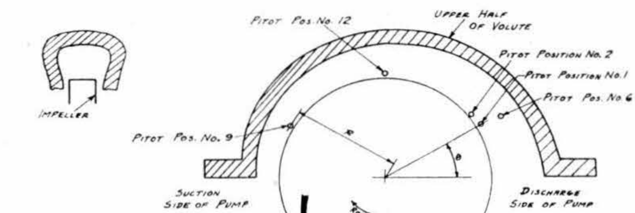
For designating the axial position of measurement across the volute, "center" means over the center of the impeller, while "right" or "left" refers to the side from this center. On the double-suction pump "right" and "left" were used with the observer facing the pump suction flange. On the single-suction pump the "right" side refers to the suction side of the pump. The test results and curves from both pumps will be given first, to be followed by a combined discussion.

Instantaneous Velocity Measurements on Byron Jackson 8-In. Double-Suction Centrifugal Pump. The pump rating and dimensions are as follows:

Capacity = 2400 gpm
Total head = 360 ft
Speed = 2500 rpm
Specific speed = 1400
Impeller outside diameter = $13^3/_8$ in.
Impeller inside width = $1^{12}/_{32}$ to $1^{13}/_{32}$ in.
Impeller outside width = $1^3/_4$ in.
Number of vanes = 8

All tests on this pump were made at 2000 rpm. The maximum efficiency at 2000 rpm was 84.6 per cent, which was the same as at the rated speed. All tests were made at plus 40 ft inlet head. Hydraulically, this pump was better than appears here. Before these pitot-tube measurements were made, this pump had received some severe treatment in previous tests, with the results that the leakage losses were increased as the efficiency decreased from an original value of 85.8 per cent to 84.6 per cent.

Fig. 9 shows the spacing of the pitot-tube stations in the volute. Since this was a horizontally split-case double-suction pump, it was not possible to provide pitot stations in the lower half of the volute, and therefore no measurements could be made in the vicinity of the tongue.



Spacing of Pitot Holes

Position	R Radial Distance	θ Degrees Counter Clockwise	Distance Pitot Center to Impeller
1	$7\frac{1}{16}$"	30.3	3/8"
2	$6\frac{15}{16}$"	37.8	1/4"
6	$8\frac{7}{16}$"	29.3	$1\frac{3}{4}$"
9	$7\frac{1}{16}$"	149.7	3/8"
12	$7\frac{1}{16}$"	90.0	3/8"

Fig. 9 Spacing of the Pitot-Tube Stations in the Volute of the Byron Jackson Pump

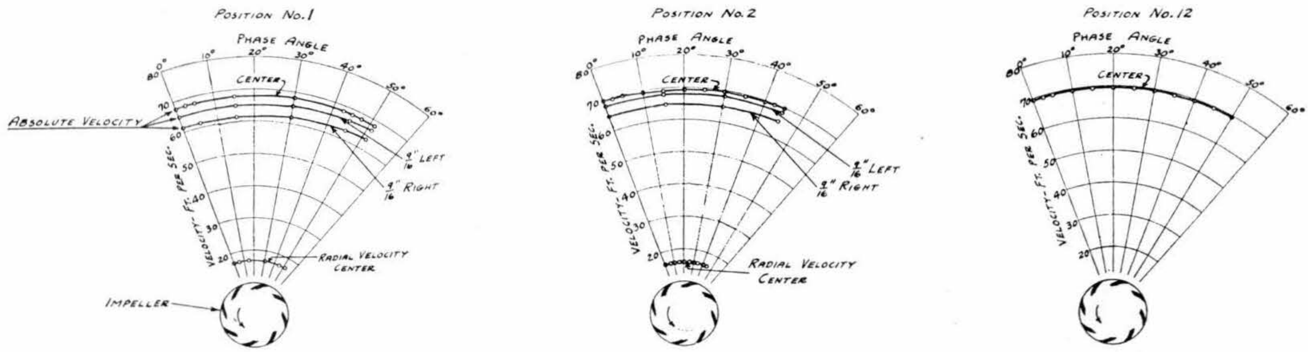


Fig. 10 Instantaneous Velocity Distribution Between Vanes in the Bryon Jackson Pump
(Pump speed = 2000 rpm, and normal capacity = 4.65 cfs.)

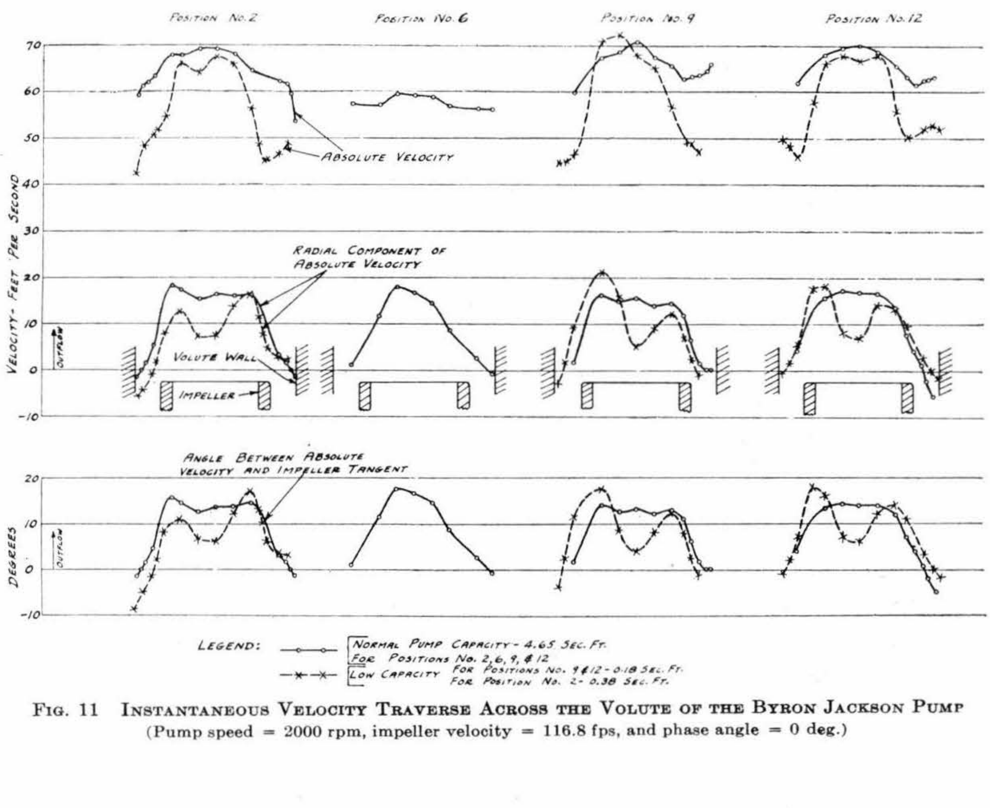


Fig. 11 Instantaneous Velocity Traverse Across the Volute of the Byron Jackson Pump
(Pump speed = 2000 rpm, impeller velocity = 116.8 fps, and phase angle = 0 deg.)

Fig. 10 shows the results of measurements to find the velocity distribution between vanes. For each set of measurements, the pitot tube was kept at a fixed position across the volute, while the phase angle was varied to traverse the impeller passage (by rotating the field of the synchronous motor driving the slide valves).

Fig. 11 shows the profiles obtained from pitot-tube traverses across the volute, each traverse being made at a constant phase angle.

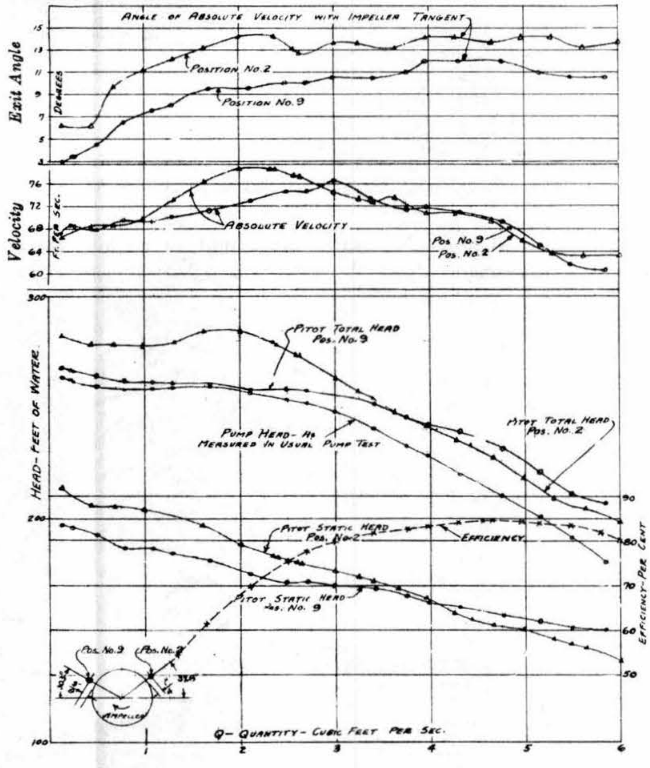


FIG. 12 PITOT-TUBE MEASUREMENTS AT DIFFERENT DISCHARGES OF THE BYRON JACKSON PUMP

Fig. 12 shows the pitot-tube measurements at different pump discharges. With the pitot at one position in the center of the volute, the pump capacity was varied.

Instantaneous Velocity Measurements on Worthington 7-In. Single-Suction Centrifugal Pump. The pump rating and dimensions are as follows:

Capacity = 2400 gpm
Head = 360 ft
Speed = 2900 rpm
Specific speed = 1720
Impeller outside diameter = $12^1/_2$ in.
Impeller inside width = $1^7/_{32}$ to $1^{15}/_{64}$ in.
Impeller outside width = $1^{17}/_{32}$ in.
Number of vanes = 7

All tests on this pump were made at 2500 rpm. The maximum efficiency at 2500 rpm was 88.6 per cent which was the same as at the rated speed. All tests were made at plus 40 ft inlet head.

Extensive measurements were made at "normal," "low," and "high" pump discharges. Normal pump discharge is that at the point of maximum pump efficiency. Low refers to a discharge of about 19 per cent of normal, while high refers to a discharge of about 142 per cent of normal.

Fig. 13 shows the spacing of the pitot-tube stations in the volute. It should be noted that there are two points of difference in the location of these stations as compared to those of Fig. 9. First, they are spaced completely around the volute, and second, they are all located at a constant radial distance from the impeller.

Fig. 14 shows the results of measurements as the pitot tube was kept at a fixed position across the volute and the phase angle varied.

Figs. 15, 16, and 17 show the profiles obtained from traverses across the volute, each traverse being made at a constant phase angle, and each figure referring to a different pump capacity. The traverses at high capacity are not complete, but the measurements are useful to some extent in a comparison with the results of tests at normal and low capacities.

Fig. 18 shows, for each pump capacity, a plot of average radial velocity vs. angle around the volute. Each point represents the average value of the corresponding profile found in Figs. 15, 16, or 17.

Fig. 19 shows the static-pressure distribution around the volute as given by the pitot tube. The static pressure plotted is that developed by the pump, and thus does not include the inlet pressure. The dashed horizontal lines indicate the mean static-pressure value for each curve.

Fig. 20 shows both the unbalanced static pressure and the momentum forces acting on the impeller. In the absence of other definite information the outside outlet width of the impeller was taken as the area over which the static pressure acts.

Fig. 21 shows the average direction of the relative exit-velocity vectors at the different pitot stations. From each traverse across the impeller width, the average radial velocity and the average tangential velocity were computed. From these two averages and the impeller peripheral velocity a relative exit angle B was calculated for each traverse.

DIMENSIONS OF PITOT LOCATIONS

POSITION NO.	"R" RADIAL DISTANCE	"θ" DEGREES	DISTANCE PITOT CENTER TO IMPELLER
A	6.70"	25.5	0.45"
B	6.68"	49.8	0.43"
C	6.68"	67.1	0.43"
D	6.66"	90.2	0.41"
E	6.67"	135.4	0.42"
F	6.68"	157.6	0.43"
G	6.69"	224.9	0.44"
H	6.68"	269.1	0.43"
I	6.72"	295.4	0.47"
J	6.71"	314.5	0.46"
K	6.71"	334.1	0.46"
L	6.71"	354.8	0.46"

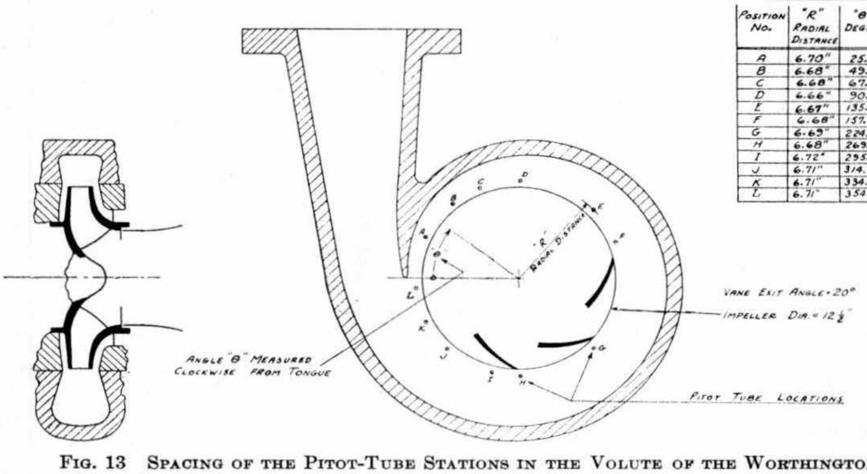


FIG. 13 SPACING OF THE PITOT-TUBE STATIONS IN THE VOLUTE OF THE WORTHINGTON PUMP

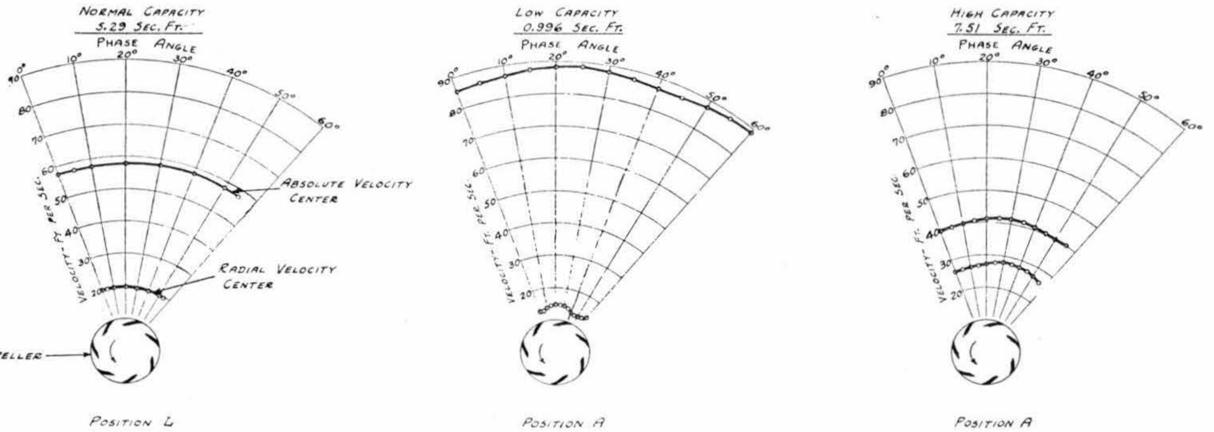


Fig. 14 Instantaneous Velocity Distribution Between the Vanes of the Worthington Single-Suction Pump at 2500 Rpm

Discussion of Results

It has been seen in the description of the test results that the measurements taken on the double-suction pump are not as complete as those made on the single-suction pump. However, in so far as they do duplicate each other, it is instructive to consider the data from both pumps simultaneously. Before doing so it should be noted that the double-suction pump has the lowest specific speed, even when both calculations are made on the same basis. If the specific speed of the double-suction machine is calculated on the basis of one half the rated capacity, as is often the practice, then it becomes even lower, i.e., it reduces to about 990, compared to 1720 for the single-suction pump.

Instantaneous Velocity Distribution in Impeller Passages. Although no measurements could be made in the impeller passages themselves, much could be inferred concerning the velocity distribution at the discharge of the impeller passages from the instantaneous measurements taken with the pitot inserted in the volute very close to the impeller. As the phase angle at which the slide valve opened was shifted, the velocity gradient between the vanes was measured. The results obtained are seen for the two pumps in Figs. 10 and 14. The most striking thing to be noticed is that there is very little velocity variation observable between the vanes at normal pump discharge. This is not in agreement with the normal conception of the impeller flow, in which it is often assumed that there is a dead water space or even a backflow along the low-pressure side of the vane. Before any conclusions are reached it is necessary to consider several characteristics of the measurements:

(*a*) The velocity distribution here obtained is not a true instantaneous picture of the flow from the volute but is rather the time variation of the velocity as the impeller passage passes a given station. To obtain the actual instantaneous velocity distribution it would have been necessary to have had a series of pitot stations spaced a few degrees apart around a portion of the impeller periphery, and to have taken a measurement with the correct phase angle at each of the stations.

(*b*) The slide valve is opened an appreciable time, i.e., about 5 deg of arc. Therefore an individual measurement is an average and not an instantaneous value, and, due to wire drawing, it is not an arithmetical average.

(*c*) Due to structural features of the pumps, clearances of from $^1/_8$ to $^5/_{16}$ in. between the impeller and the pitot were necessary. Some change of velocity could therefore occur between the points of discharge and measurement. For example, this together with (*b*) explains why the flow is not zero during the time the vane itself is passing the measuring point.

After taking all of the foregoing factors into consideration, the conclusion is still unavoidable that in high-efficiency pumps, operating under conditions of normal discharge, the velocity distribution across the impeller passage discharge is surprisingly uniform.

Fig. 14 shows that for capacities either above or below normal, some velocity gradients are observable. However, in no case are they as great as previous studies have indicated.

Velocity Profiles Across the Volute. Figs. 11, 15, 16, and 17 show the velocity profiles obtained by making traverses across the volutes from wall to wall at the different stations. It will be noted immediately that there are no marked breaks in the profiles to show the locations of the impeller shrouds. In this connection no attempt should be made to find zero flow between the shroud and the case at any single pitot station. Although the total flow across this space must be equal to the leakage through the wearing rings, it is very possible to have flow into the space in one region of the volute and out in other regions. In fact, this circulation can act as an energy pump by entering this space from the volute at a relatively low velocity and later returning to another section of the volute with a higher velocity; For example, in Fig. 15 the radial-velocity components show a net inflow at relatively low velocity to this space from stations *A* to *G* and outflow at higher velocity from station *H* to *L*. It will be remembered that originally pump volutes were built with small clearances between the walls and the impeller periphery, but that efficiencies were improved when the clearances were made much larger. The energy flow previously mentioned may account for some of this improvement, because, with the close clearances, the energy imparted to the fluid in these spaces by disk friction on the shrouds is trapped in the spaces and must be dissipated without benefit to the pump performance, while with ample clearances at least a part of this energy may be carried out into the volute and utilized.

An inspection of the radial-velocity distributions for the low-capacity readings in Fig. 11 shows that under this condition the flow has two high-velocity peaks at each station. An obvious suggestion is that this is due to the double-suction impeller, which is fundamentally two impellers placed back to back. However, this is immediately seen to be erroneous when the same peaks are found in Fig. 16, which is plotted from measurements of the single-suction pump. The most reasonable explanation of these peaks appears to arise from a consideration of the centrifuge action of the shroud. Professor von Kármán has suggested a calculation to help explain this matter. In the following calculation no claim is made to express exactly the

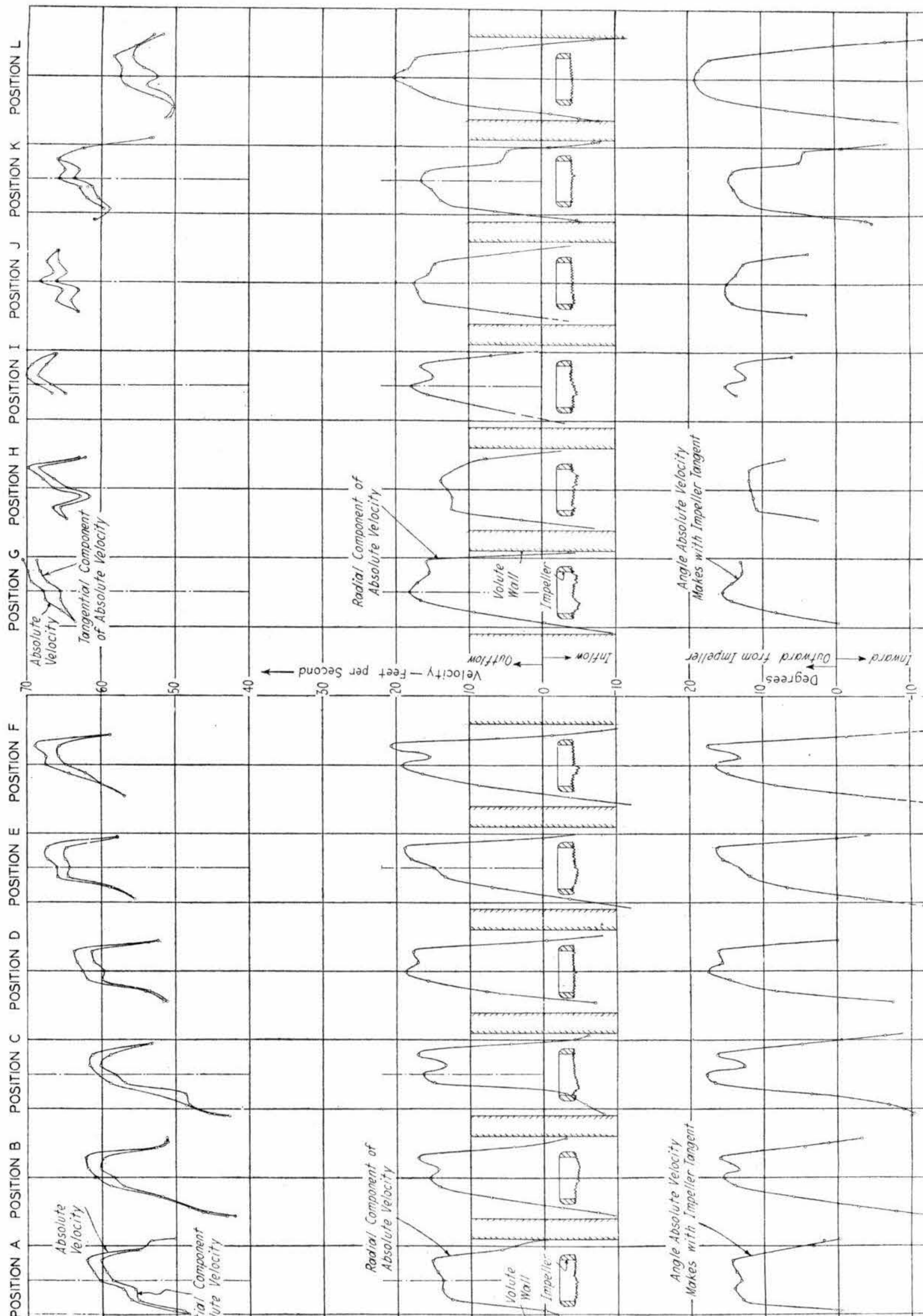


Fig. 15 Velocity Traverses Across Volute in the Worthington Pump

(Pump speed = 2500 rpm, phase angle = 20 deg, pump head = 274 ft, impeller peripheral velocity = 136.3 fps, and normal capacity = 5.29 cfs.)

FIG. 16 VELOCITY TRAVERSES ACROSS VOLUTE IN THE WORTHINGTON PUMP FOR LOW CAPACITY
(Pump speed = 2500 rpm, phase angle = 20 deg, pump head = 336 ft, impeller peripheral velocity = 136.3 fps, and low capacity = 0.996 cfs. Note change of scale between positions F and G.)

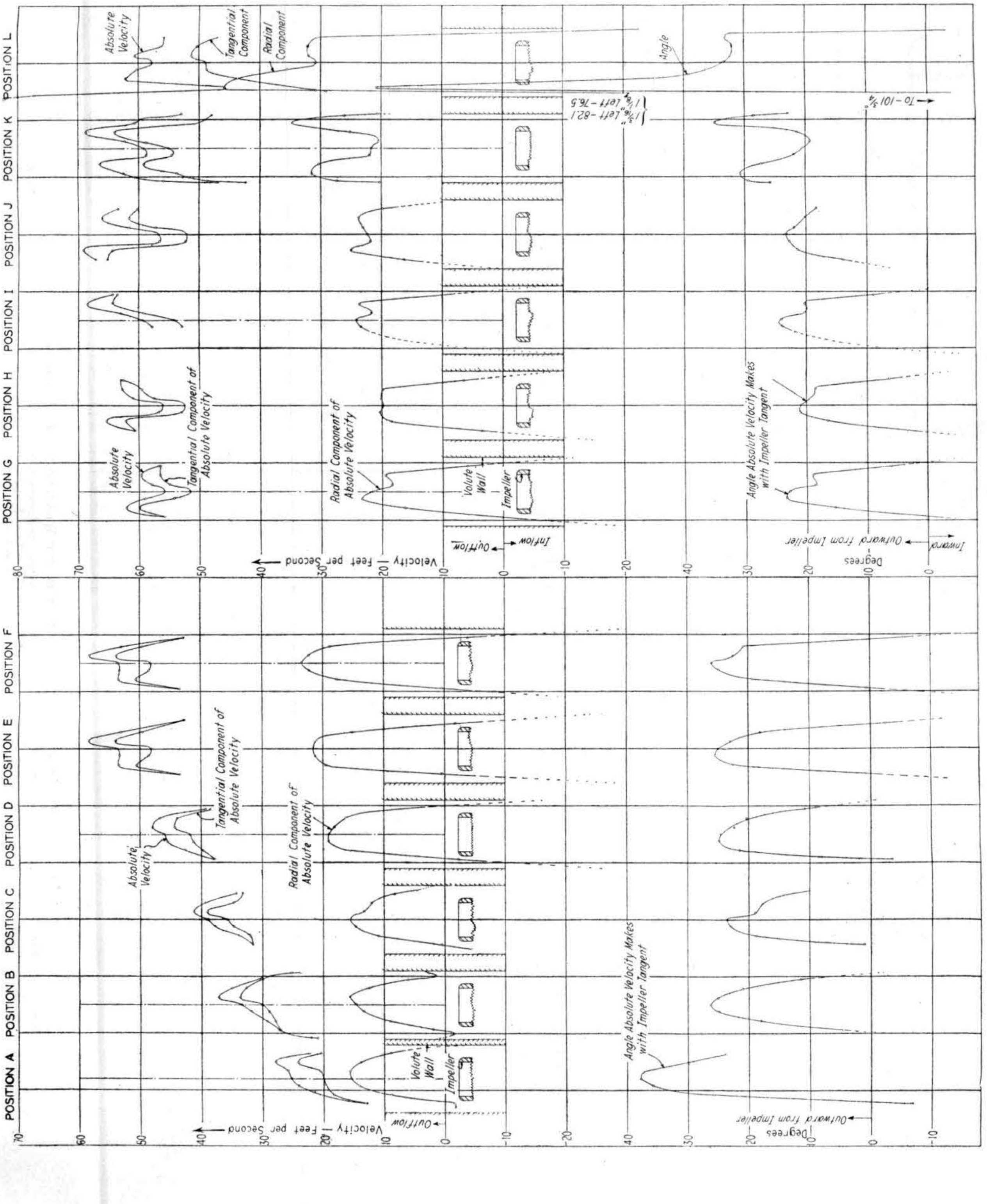
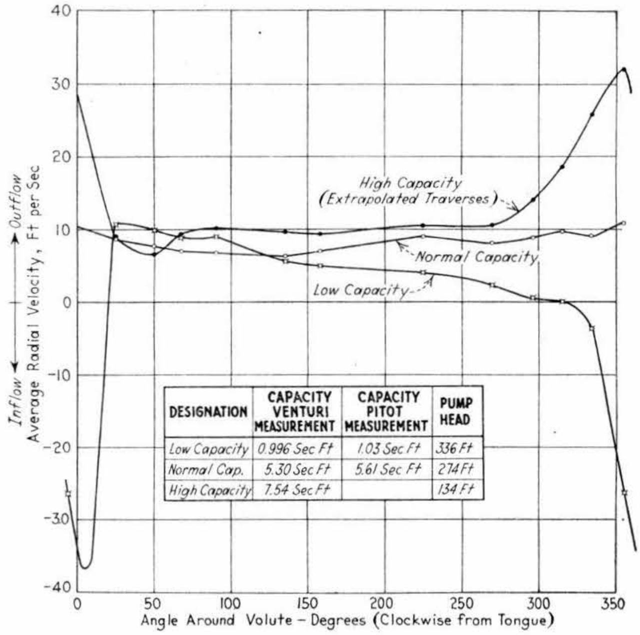


Fig. 17 Velocity Traverses Across Volute in the Worthington Pump for High Capacity

(Pump speed = 2500 rpm, phase angle = 20 deg, pump head = 134 ft, impeller peripheral velocity = 136.3 fps, and high capacity = 7.54 cfs. Note change of scale between positions F and G.)

complicated conditions in a pump, but the computation serves to give the order of magnitude of the peaks. Professor von Kármán (21) has treated the problem of the frictional resistance of a rotating disk for the case of turbulent flow. He considered a smooth flat disk wetted on one side. The various momentum changes were taken into account, and the velocity distribution in



DESIGNATION	CAPACITY VENTURI MEASUREMENT	CAPACITY PITOT MEASUREMENT	PUMP HEAD
Low Capacity	0.996 Sec Ft	1.03 Sec Ft	336 Ft
Normal Cap.	5.30 Sec Ft	5.61 Sec Ft	274 Ft
High Capacity	7.54 Sec Ft		134 Ft

FIG. 18 AVERAGE RADIAL VELOCITY AT DIFFERENT ANGLES AROUND THE VOLUTE OF THE WORTHINGTON PUMP

(Pump speed = 2500 rpm, and impeller peripheral velocity = 136.3 fps.)

the boundary layer was assumed to follow the seventh-root law. The following expressions were derived in the treatment

$$C_o = 0.162\ R\omega$$

$$\delta = 0.522\ R \left(\frac{\nu}{R^2\omega}\right)^{1/5}$$

where C_o = maximum radial velocity in the boundary layer, R = radius of disk, ω = angular velocity of the disk, δ = thickness of the boundary layer, and ν = kinematic viscosity of the fluid. Applying the two relations to the two pumps under consideration, and taking for R the radii of the peripheries of the impellers, the following values are obtained: For the double-suction pump C_o = 18.9 fps, and δ = 0.15 in. For the single-suction pump C_o = 22.1 fps, and δ = 0.14 in.

Figs. 11 and 16 show that some of the measured radial-velocity peaks are close to these computed values of C_o. Thus it is indicated that each shroud acts as a centrifuge to discharge a sheet of water into the volute.

The question might be raised as to why there is a lack of a "shadow" in the velocity profiles above the shroud. In both pumps the shrouds are about $^3/_{16}$ in. thick, and in this space there should be no radial flow. However, it is quite possible that in the short radial distance (between the impeller periphery and the measuring tube) the flows from both sides could diverge, and that these divergencies could combine to give an appreciable positive velocity over the shroud thickness.

The difference in the velocity of these two flows, together with their initial separation due to the shroud thickness, offers a possible explanation of the shift of the velocity peaks away from the computed boundary layer and toward the center of the impeller. Referring again to Figs. 15, 16, and 17, it will be seen that there are unsymmetrical peaks on the absolute-velocity profiles. For both normal and high capacities, these peaks are on the suction side of the impeller center, while for low capacity the peaks are on the shaft side. It would be interesting to observe whether or not this shift of the position of the peaks occurred at the same time as the shift in the direction of the thrust commonly observed in single-suction pumps.

Pitot-Tube Measurements for Variable Pump Capacity. Fig. 12 shows the comparison between the head developed by the double-suction pump and the corresponding static and total heads as measured at two stations in the volute, for a wide range of capacities. Although the measurements were taken with the pitot fixed at the center line of the impeller, the static-head readings

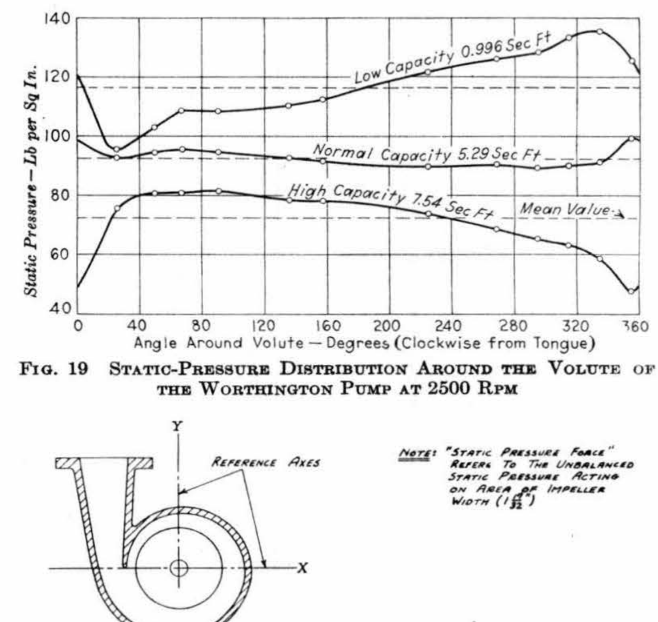


FIG. 19 STATIC-PRESSURE DISTRIBUTION AROUND THE VOLUTE OF THE WORTHINGTON PUMP AT 2500 RPM

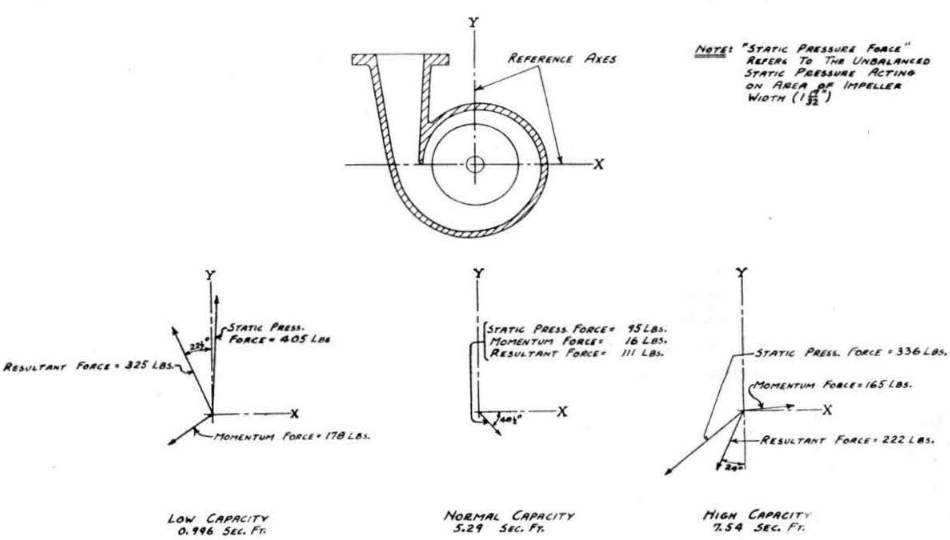


FIG. 20 UNBALANCED RADIAL FORCES ACTING ON THE IMPELLER OF THE WORTHINGTON PUMP

probably represent average values across the width of the volute, since it was found that the static pressure was relatively constant throughout the traverse. On the other hand, the total-head readings are not so representative, as shown by the absolute-velocity traverses of Fig. 11. The difference between the static pressures at the stations shows that there may be a possibility of an unbalanced radial force on the impeller, especially in the low-capacity region. Since the two curves cross in the vicinity of normal capacity, it might be expected that the direction of the unbalanced force would reverse in the high-capacity region.

Radial-Velocity Distribution Around the Volute. The complete ring of pitot stations provided in the single-suction pump has made it possible for the first time to secure sufficient data to compute the radial-velocity distribution around the entire volute. Fig. 18 shows the three distribution curves obtained.

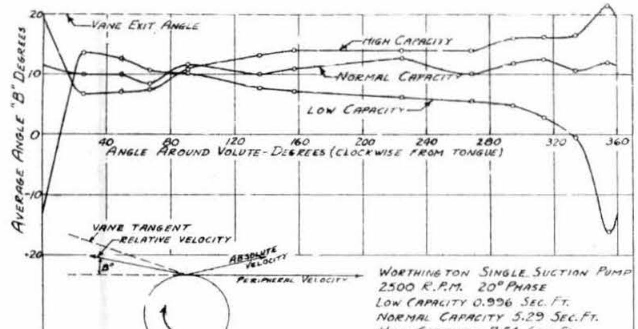


FIG. 21 DIRECTION OF AVERAGE RELATIVE EXIT VELOCITY AT DIFFERENT PITOT STATIONS AROUND THE VOLUTE OF THE WORTHINGTON PUMP

The first item to note is that the three capacities show markedly different characteristics. For normal capacity, the radial velocity is relatively constant around the entire volute, although it is by no means exactly so. At high capacity there is a region of uniform velocity extending over about 200 deg. The remainder of the circumference, which is in the vicinity of the tongue, shows a much higher value of outflow. At low capacity there is no region of uniform flow. However in the region of the tongue there is a very striking zone of high inflow.

Both high- and low-capacity distributions lead to the same conclusion that, except for normal-capacity operation, there is always a large velocity variation in the impeller passage each revolution. Under some low-capacity conditions this becomes an actual reversal of flow. Probably this velocity fluctuation accounts for some loss in pump efficiency.

Unbalanced Radial Forces. In the operation of high-head pumps, trouble sometimes arises with the wearing rings, caused by large shaft deflections. This results in metallic contact excessive wear and, hence, increased leakage. In severe cases, shafts have been known to break due to fatigue. Fig. 19 is a plot of the static-pressure distribution around the impeller of the single-suction pump, in an attempt to study this unbalanced force. Lack of uniformity of static pressure would of course give rise to such a radial resultant force. Note that at normal capacity the static pressure is quite uniform, while very wide variations are present for both low and high capacities.

If the radial forces represented in Fig. 19 are added vectorially, the magnitude and direction of the unbalanced resultant is obtained. Reference to Fig. 20 shows that the directions of the resultants are quite different for high- and low-capacity operation. This could have been predicated from the difference in shapes of the corresponding pressure distributions of Fig. 19.

This force resulting from the unbalanced static-pressure distribution is, however, not the only radial force acting on the impeller. A nonuniform velocity distribution will give rise to an unbalanced momentum force in the same manner. This must be added to the static-pressure force to obtain the total hydraulic reaction.

The analysis of the three operating conditions presented in Fig. 20 shows that at low capacity the deflection may be about three times that at normal capacity, while for high capacity it may be twice that at normal. The latter is not so serious, since operation at high capacity is not always necessary. On the other hand, the low-capacity range is always passed through when the machine is started and stopped, and continuous operation in this region is not uncommon.

Measurements of the deflections at the impeller wearing ring during operation under the various indicated capacities have shown that the impeller movement was in a direction which agrees with that of the resultant force vectors as determined in Fig. 20. A study of the stress-strain conditions in the shaft agreed in magnitude with those calculated from the hydraulic reaction, although in general the former are somewhat higher. However, this stress-strain analysis involves considerable difficulty because of the uncertainty as to the amount of the bearing and casing deflections.

Direction of Relative Exit Velocity From Impeller. A question of considerable interest is the relation between the vane exit angle and the direction of the relative velocity of the fluid leaving the impeller. One item to note is that the relative exit angle is less than the vane angle for all points except one. Again, at normal capacity the conditions are relatively uniform around the entire perpihery, while at low and high capacity there is a wide deviation between the different stations. The averages given in Fig. 21 show that at low capacity the deviation between the vane angle and the relative velocity is 14.2 deg, for normal capacity 9 deg, and for high capacity 6.8 deg. Note that in the vicinity of the tongue the deviation reaches as much as 36 deg.

SUMMARY OF RESULTS

1 There is practically no instantaneous velocity variation in the impeller discharge at normal capacity and only slightly more at low or high capacities during the time of passage of one vane space past a measuring station.

2 There is a strong circulation between the volute and the impeller clearance space which apparently acts as an energy pump and helps to minimize losses.

3 For low-capacity conditions, double peak-velocity profiles were found, probably due to a centrifuge action of the shroud.

4 For normal capacity, the radial-velocity distribution around the volute is relatively uniform, while for high or low capacities large variations are found.

5 For low capacity, a high ratio of inflow is observed in the region of the tongue, while for high capacity a high outflow occurs in the same area.

6 Nonuniform velocity and static-pressure distribution combine to produce unbalanced radial forces on the impeller. Maximum values exist during low-capacity operation, while the forces are at a minimum for normal rates of discharge.

7 A considerable variation in the deviation between the vane exit angle and the relative velocity was observed. The average deviation was greatest for low-capacity conditions and least for high-capacity conditions. At normal discharges it was 9 deg.

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